MEMORANDUM
INTERMOUNTAIN POWER SERVICE CORPORATION
INTERMOUNTAIN GENERATING STATION

MEMO BY: Wes Bloomfield TO: Phil Tice

DATE: November 3, 1987 FILE NO:

SUBJECT: Follow up Report Concerning the Cooling Tower Fan Blade Resonance and Ceramic Cooling Towers Proposed use of a Ten Blade Fan System

REFERENCE: Letter, dated January 29, 1987, Resonance of Cooling Tower Fan Blades may be causing fan failures.

Several months ago a very sophisticated vibration test was performed on cooling tower fan 1A03. The results of the test showed that the cell stack design is exciting the blades at their natural resonance component. Ceramic's proposed use of a ten blade fan system will not effect this resonance problem, however the overall influence may be lessened.

Fan cell 1A03 was selected for this special vibration test because of several tripping problems due to a blade droop switch being activated. Acceleration probes were mounted at the end of three of the seven blades, (see picture #1). A cable was then run from the probe to a recorder and power system, which were mounted in the center of the fan hub, (see pictures 2 thur 5). A key phaser was also attached to the fan hub and referenced to a point on the gear reducer. With this unique setup, vibration data concerning the blade movement could be obtained while the fan was in operation.

The data gathered first was the result of a fan static condition, while air was short circuited through the dormant cell. As attachment 1 indicates the three blades were being excited at natural blade resonance, 412 cpm. Attachment two indicates the signal waveform of the three blades as the fan was in operation. The waveforms have been integrated from acceleration to velocity with the ADRE 3 Data Acquisition System. All values indicated in these plots should be multiplied by a factor of 10. Attachment 2 indicates very clearly that each blade was being excited four times during each revolution. Other minor spikes in the waveform indicate reaction to other blades moving in the fan system, thus creating a fan hub wobble. Signature traces, cascade plots, and careful attention to the key phase reference mark in attachments 3 - 6, bear this last comment out. The probe on blade # 5 was damaged shortly into the test, so data was not available.

Mr. Alan Thompson
Bently Nevada Corporation
PO 157
Minden, Nevada 89423

Subject: Vibration Article

Dear Alan:

Please find attached a case history which demonstrates the use of the ADRE 3 Data Acquisition System. Please refrain from using any company name i.e. Ceramic, Hudson, or Intermountain Power Service Corporation, you may however use my name if desired. I appreciate this opportunity and I hope the data supplied will present an interesting discussion for your readers. Before the article is published, I would like the opportunity to review the discussion if at all possible. Should you require any further information please call me at (801)864-4414 extension 6483.

Sincerely,

Wes Bloomfield

Wes Bloomfield
Reliability Engineer

attachments
cc:PBT

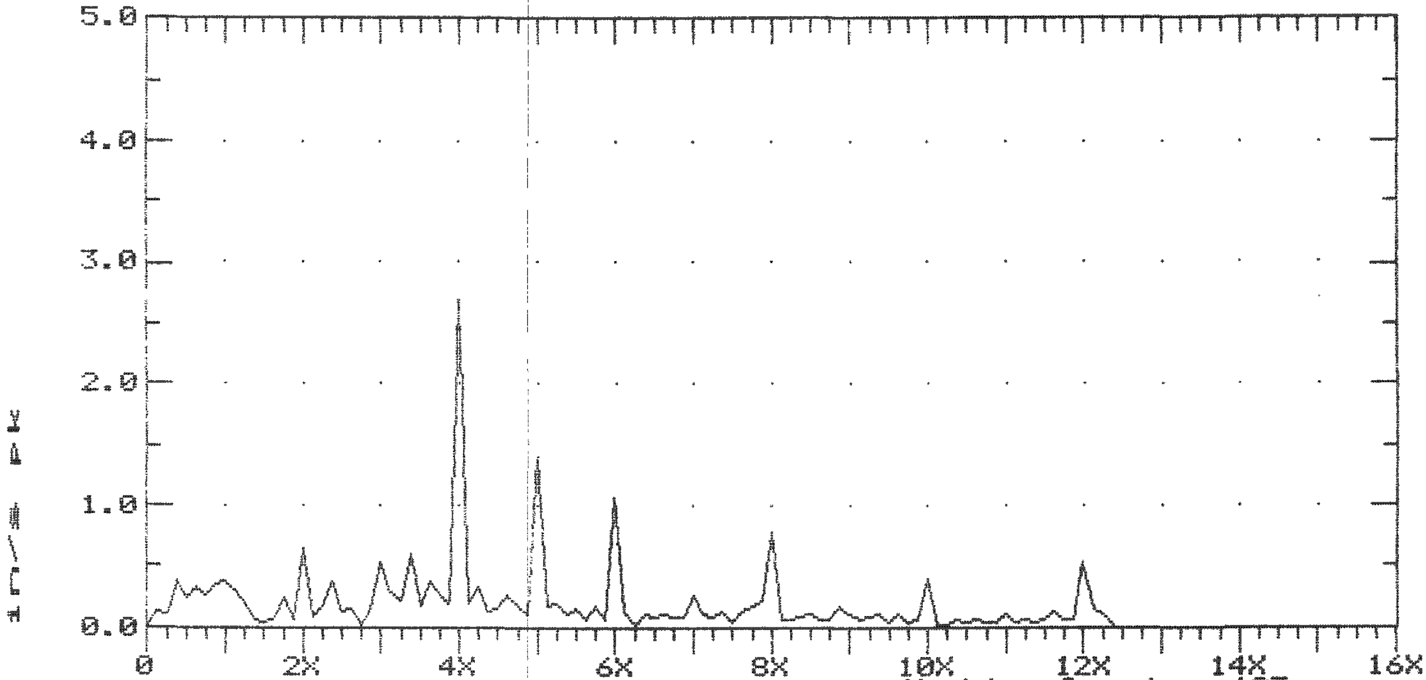
P.S. If necessary, the use of the company name, IPSC may be possible with proper approval from the appropriate trusties.

ATTACHMENT # 4

COMPANY : IPSC ENGINEERING
PLANT : UNIT 1
JOB REFERENCE: Cooling Tower Fans
MACHINE TRAIN: 1A03 Clg Twr Fan

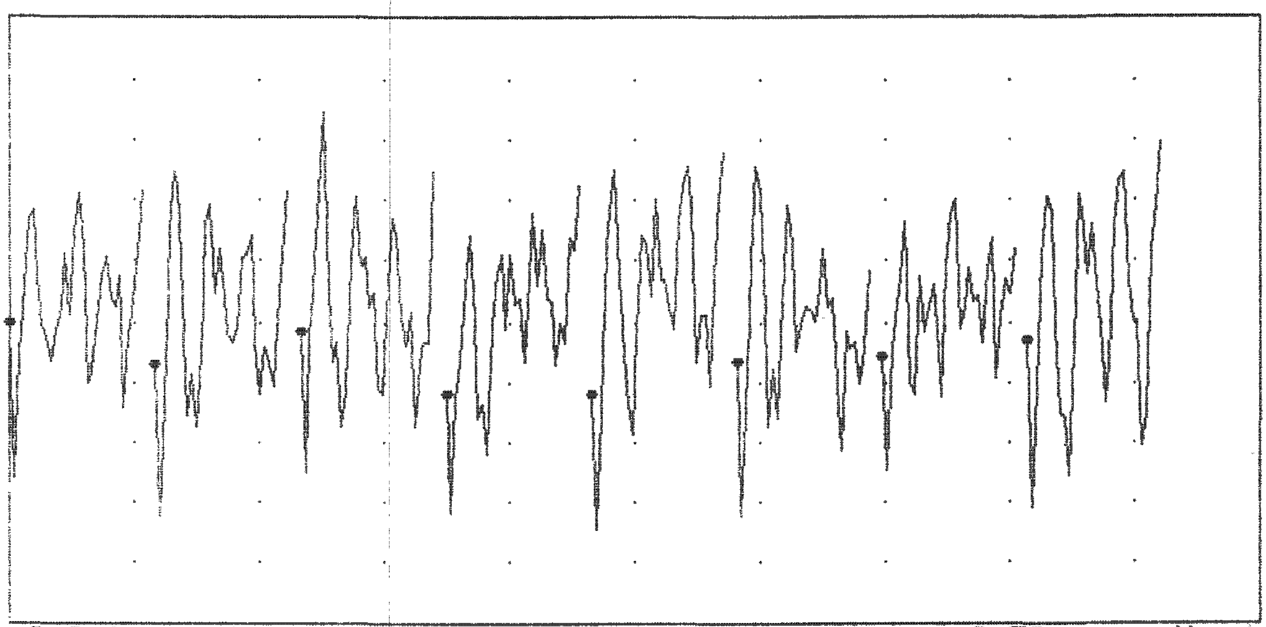
PLOT No.______

Machine: Cooling Tower Fan CH# 2 Blade # 3
16 OCT 87 08:20:49.8 UNCOMP



Machine Speed = 103 rpm

Machine: Cooling Tower Fan CH# 2 Blade # 3
16 OCT 87 08:20:49.8 Steady State UNCOMP



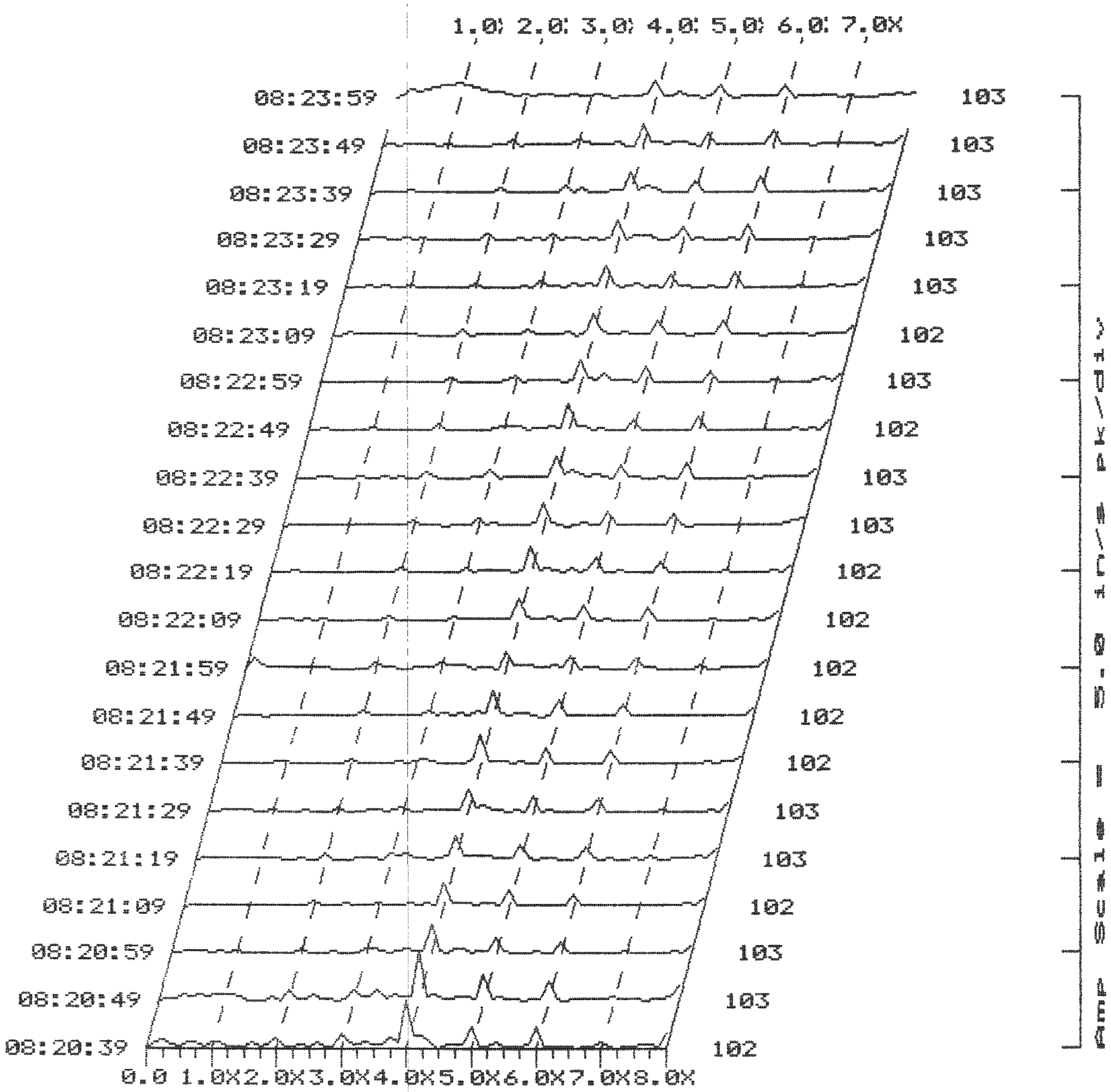
2.0 in/s /div CW Rotation 103 rpm 0.50 sec/div

ATTACHMENT # [illegible]

PLOT No.______

COMPANY : IPSC ENGINEERING
PLANT : UNIT 1
JOB REFERENCE: Cooling Tower Fans
MACHINE TRAIN: 1A03 Clg Twr Fan

MACHINE: Cooling Tower Fan CH# 2 Blade # 3
16 OCT 87 08:20:39.8 to 16 OCT 87 08:23:59.8 Steady State UNCOMP



After this last test was performed, Ceramic removed any blades which were considered to be at risk (blades with wrinkles in the neck). Since this work was performed, there has not been another failure on Unit 1. If failures should again begin to occur, then blades should be stiffened with the addition of foam to the inter void. This has been determined by Ceramic and Hudson Fan as the most economical and effective fix for the problem.

Ceramic's proposed use of a ten blade fan system to improve performance will not effect the blade resonance. The addition of three blades will however, lesson the reaction of the other blades moving in the cell. This occurs because a blade losing load will be counteracted by a blade on the opposite side of the fan losing load at the same time. Thus the overall fan wobble will be lessened. Signatures gathered from cells 2A09, 2A10, and 2A12, newly installed ten blade systems, bear this out.

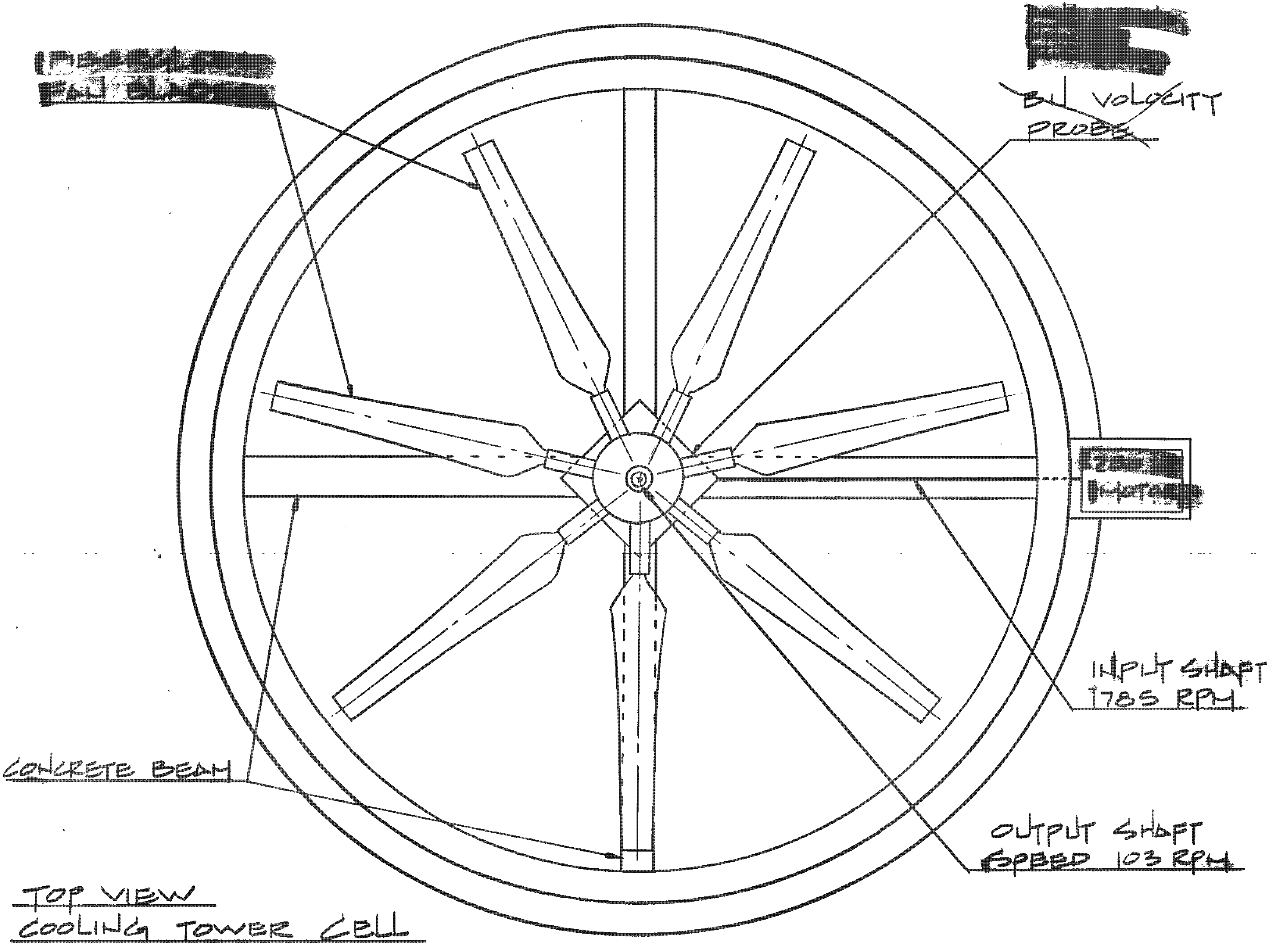
In Conclusion, the blade resonance problem referenced in earlier reports has been substantiated as being a contributor to the blade failures. If more failures should occur, then stiffening the blades by the addition of foam in the inter void of the blade should correct the problem. Ceramic's use of the ten blade system will have no effect on the blade resonance but will lesson the fan hub wobble experienced by the seven blade system. If you should require further information please contact me at extension 6483.

Sincerely,

Wes Bloomfield

Wes Bloomfield
Reliability Engineer

attachments



TOP VIEW
COOLING TOWER CELL

TEN BLADE FAN SYSTEM

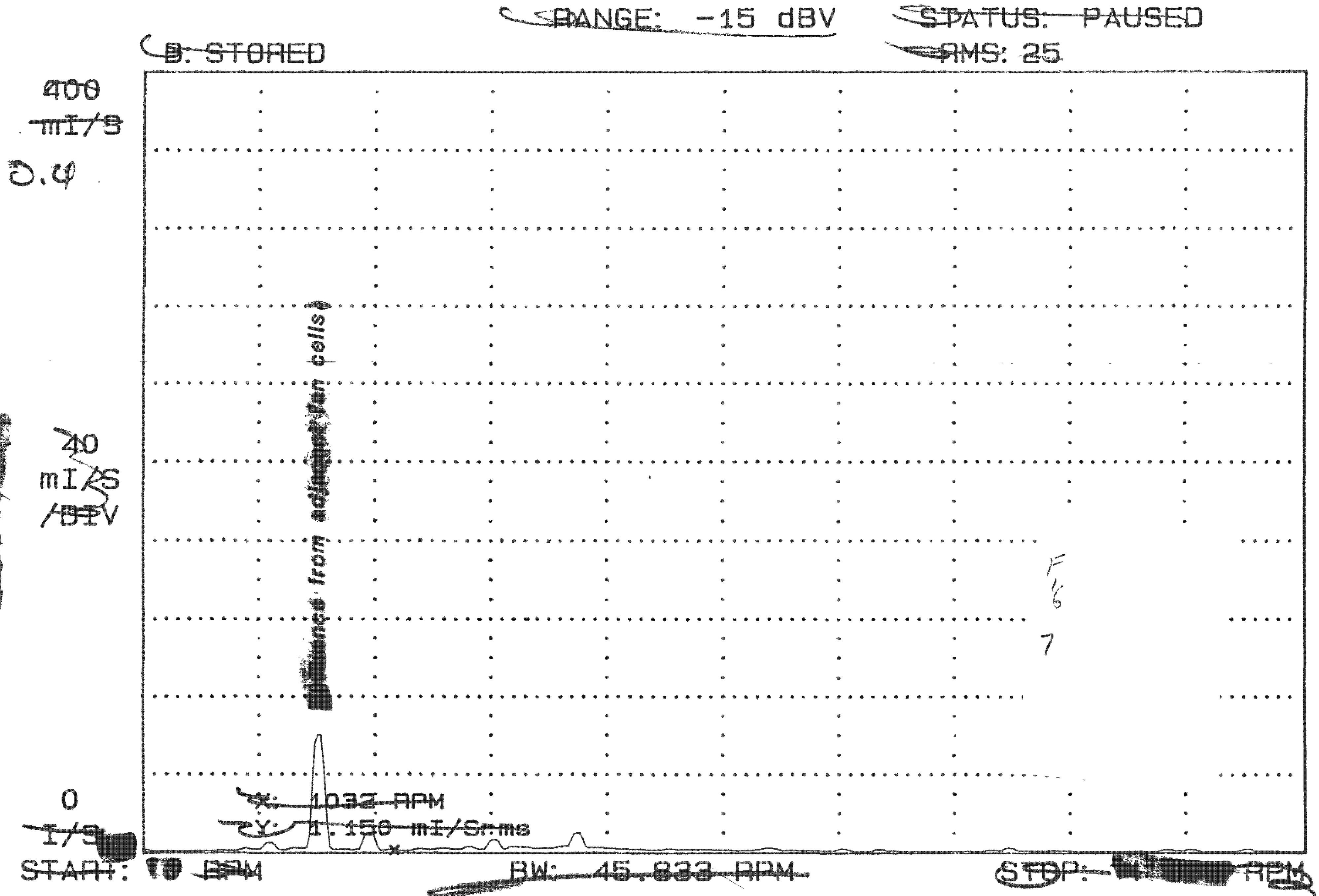


Figure 7

SEVEN BLADE FAN SYSTEM

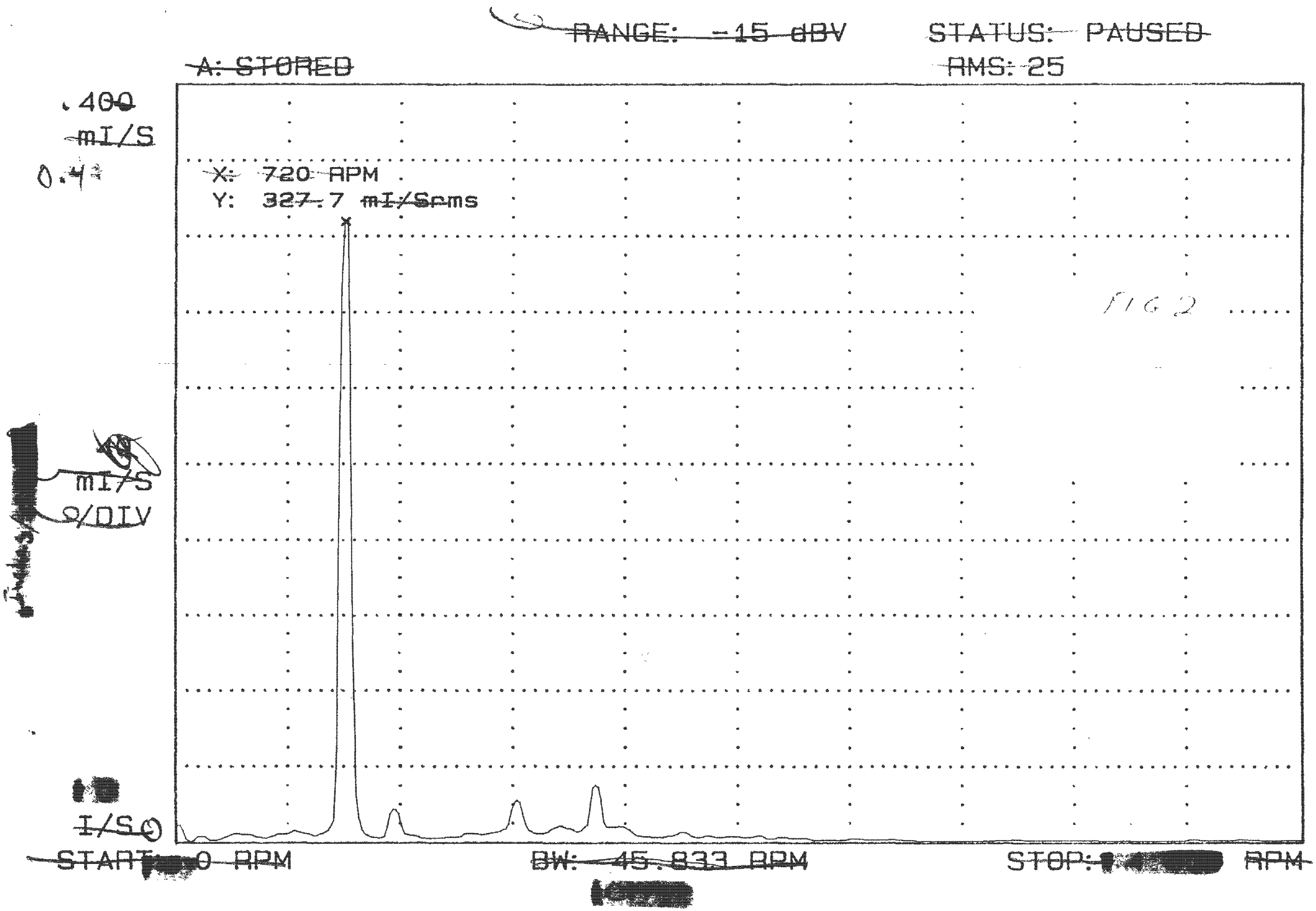


Figure 2

IP12_011519

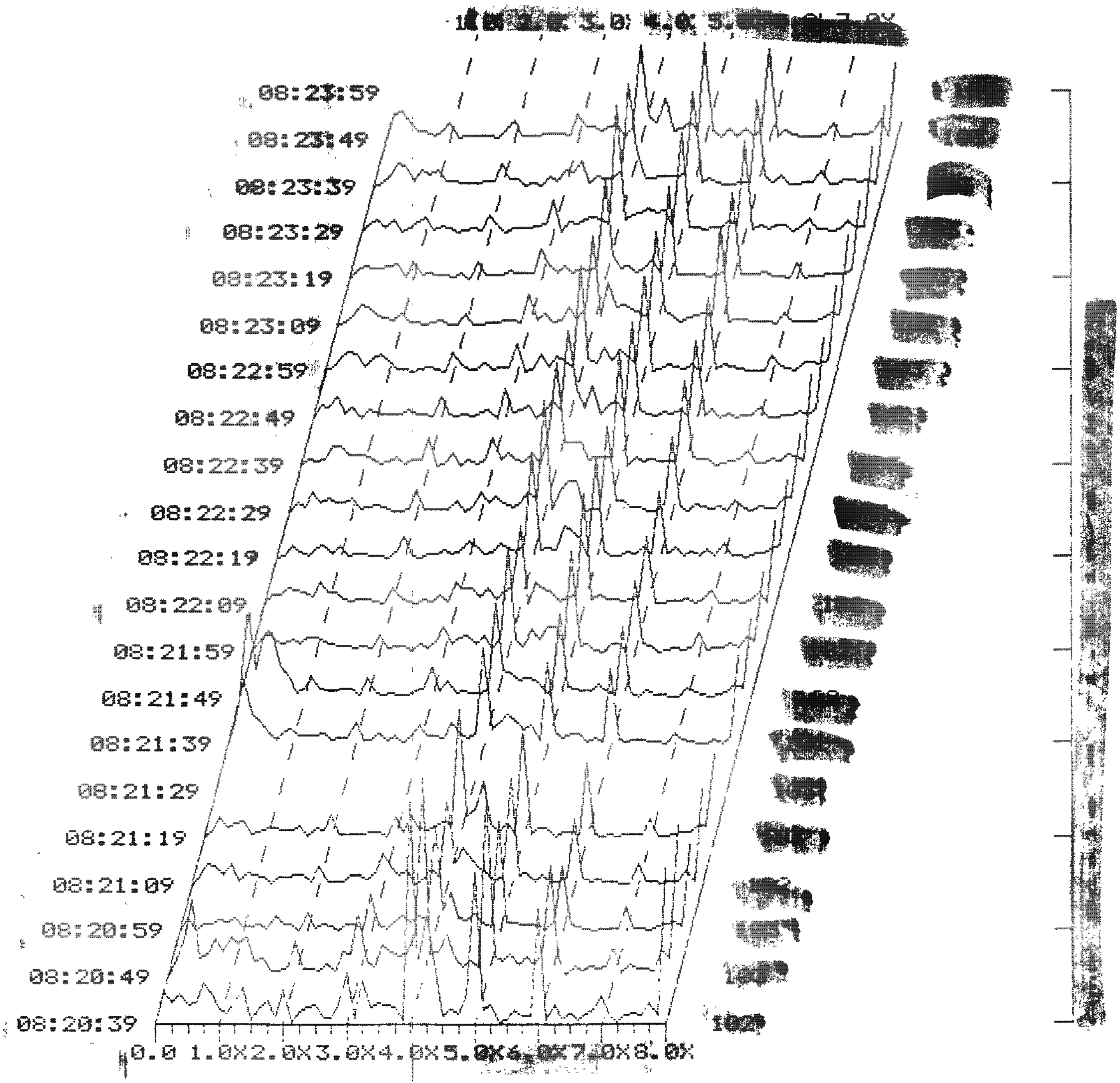
Fig 4

ATTACHMENT # 5

COMPANY : IPSC ENGINEERING
PLANT : UNIT 1
JOB REFERENCE: Cooling Tower Fans
MACHINE TRAIN: 1A03 Clg Twr Fan

PLOT No. ______

MACHINE: Cooling Tower Fan [redacted]
16 OCT 87 08:20:39.8 to 16 OCT [redacted]



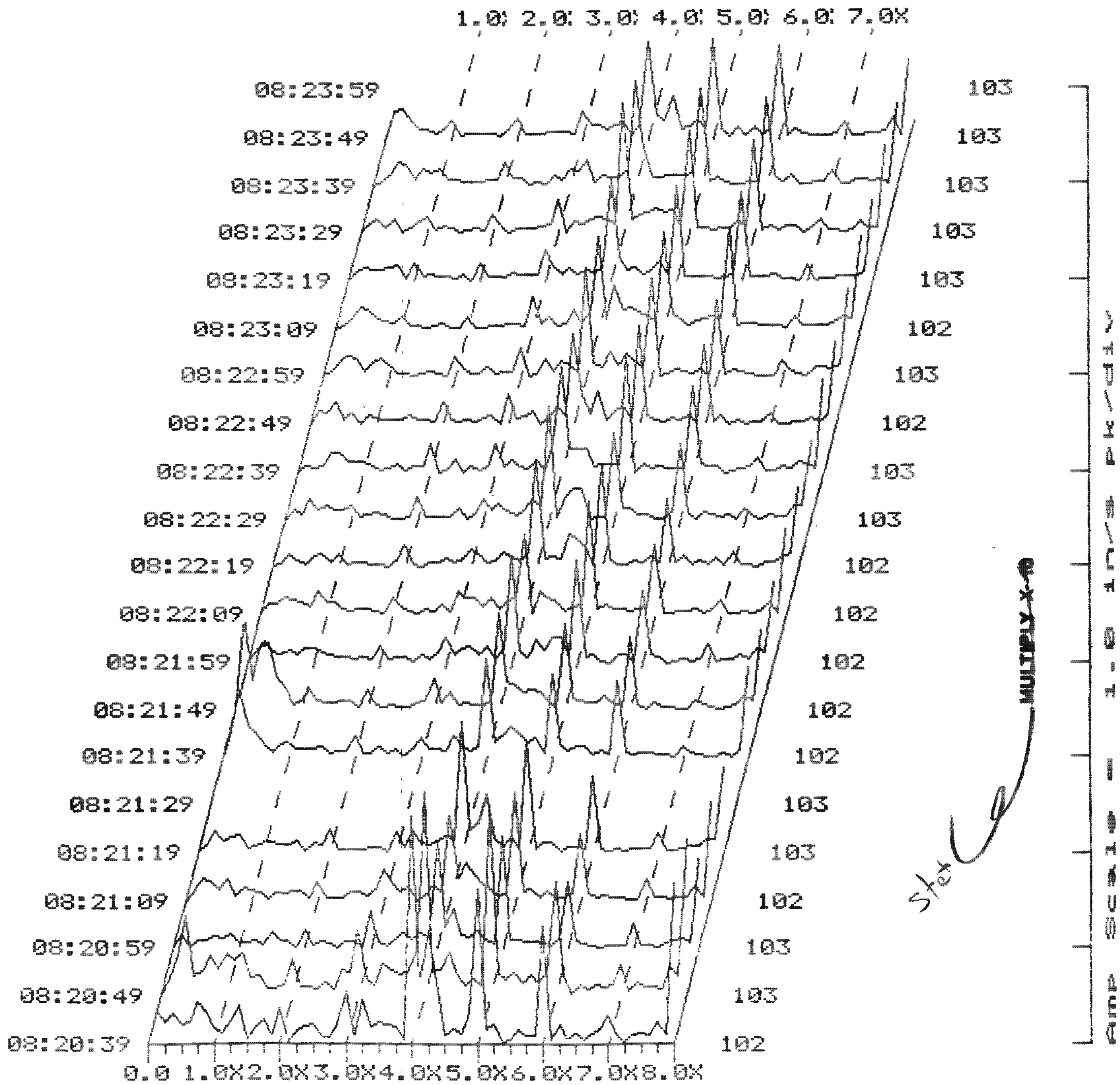
IP12_011520

FIG 4

5

COMPANY : IPSC ENGINEERING
PLANT : UNIT 1
JOB REFERENCE: Cooling Tower Fans
MACHINE TRAIN: 1A03 Clg Twr Fan

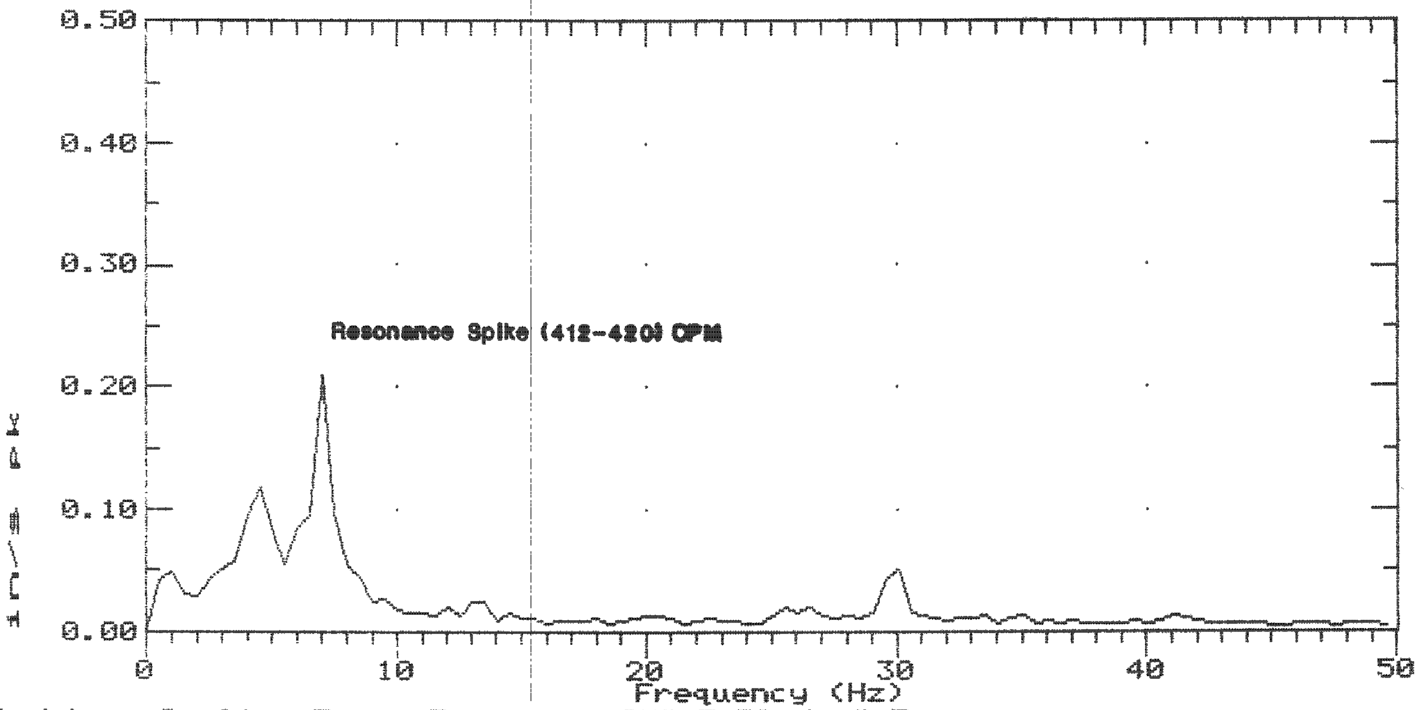
MACHINE: Cooling Tower Fan CH# 1 Blade # 1
16 OCT 87 08:20:39.8 to 16 OCT 87 08:23:59.8 Steady State UNCOMP



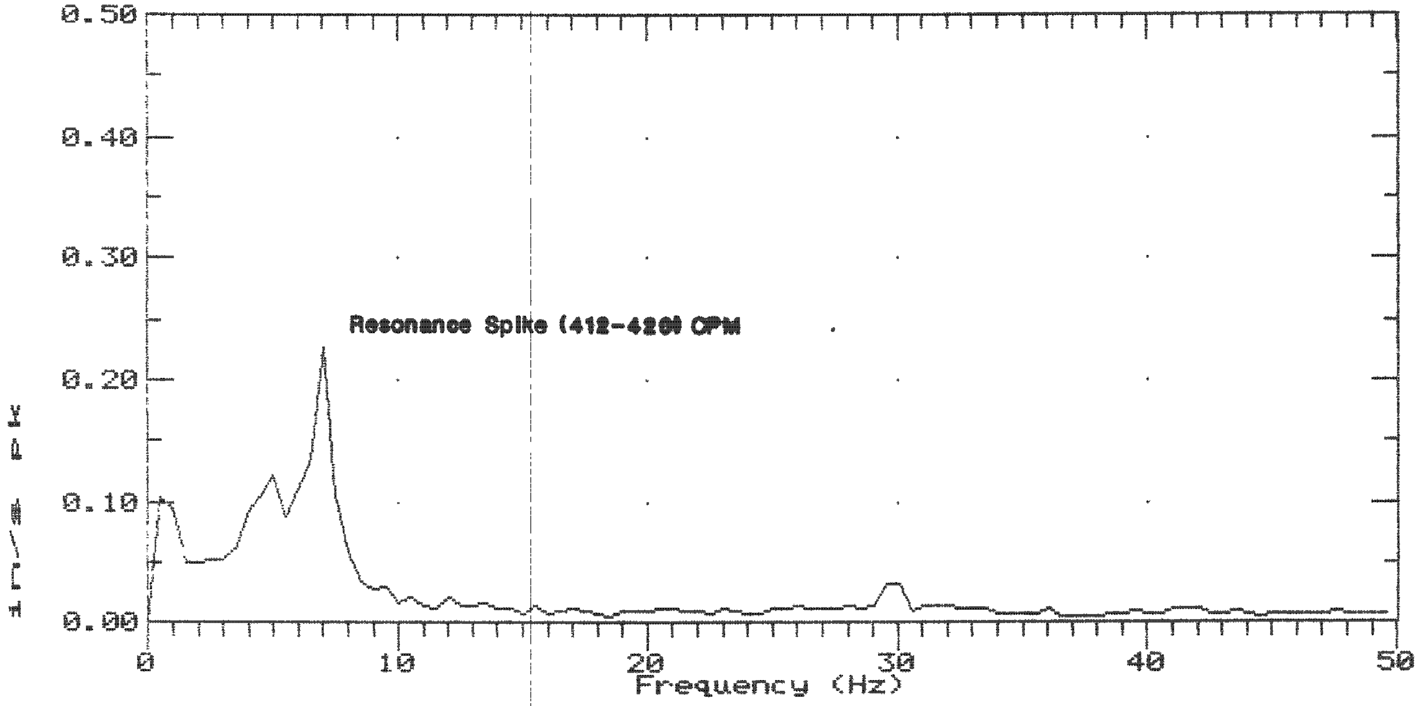
COMPANY : IPSC ENGINEERING
PLANT : UNIT 1
JOB REFERENCE: Cooling Tower Fans
MACHINE TRAIN: 1A03 Clg Twr Fan

PLOT No.______

Machine: Cooling Tower Fan CH# 1 Blade # 1
16 OCT 87 08:14:56.9 to 16 OCT 87 08:15:46.9 UNCOMP # Ave. = 6 (spec)



Machine: Cooling Tower Fan CH# 3 Blade # 5
16 OCT 87 08:14:56.9 to 16 OCT 87 08:15:46.9 UNCOMP # Ave. = 6 (spec)

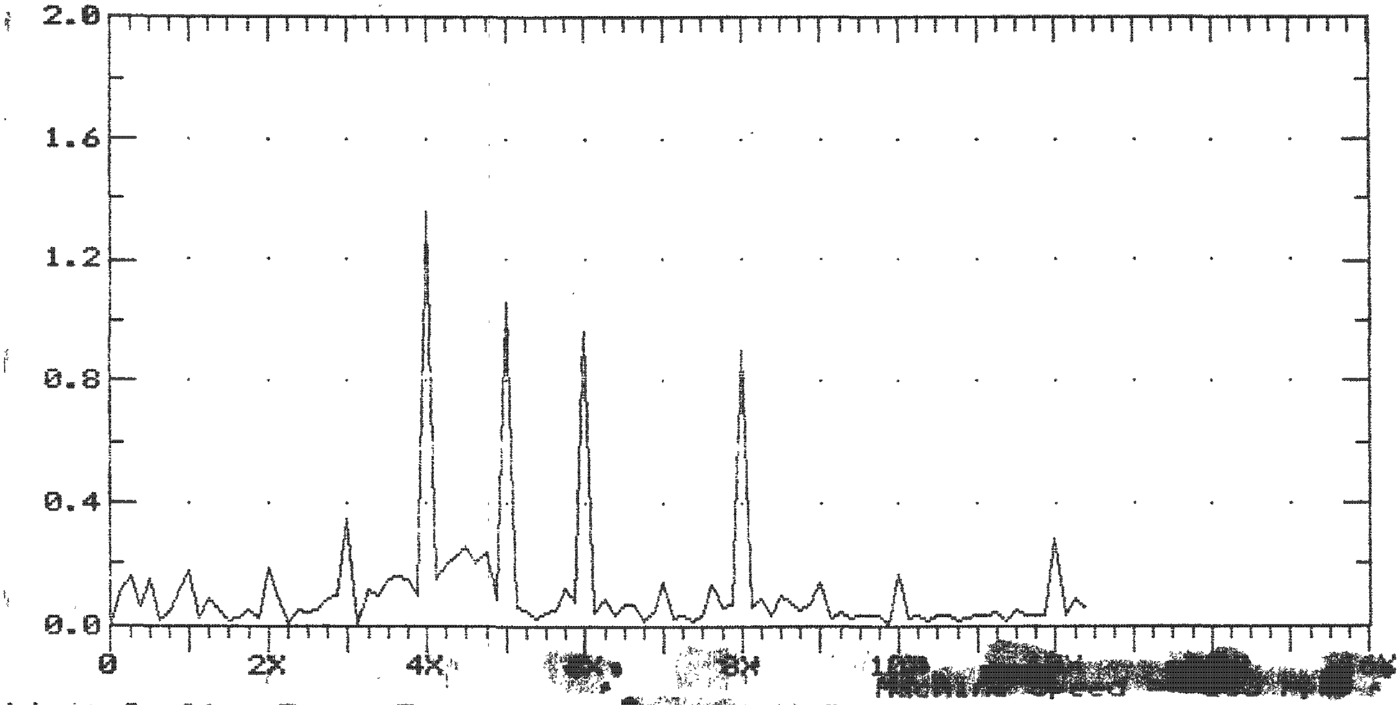


ATTACHMENT # 8

COMPANY : IPSC ENGINEERING
PLANT : UNIT 1
JOB REFERENCE: Cooling Tower Fans
MACHINE TRAIN: 1A03 Clg Twr Fan

PLOT No. ______

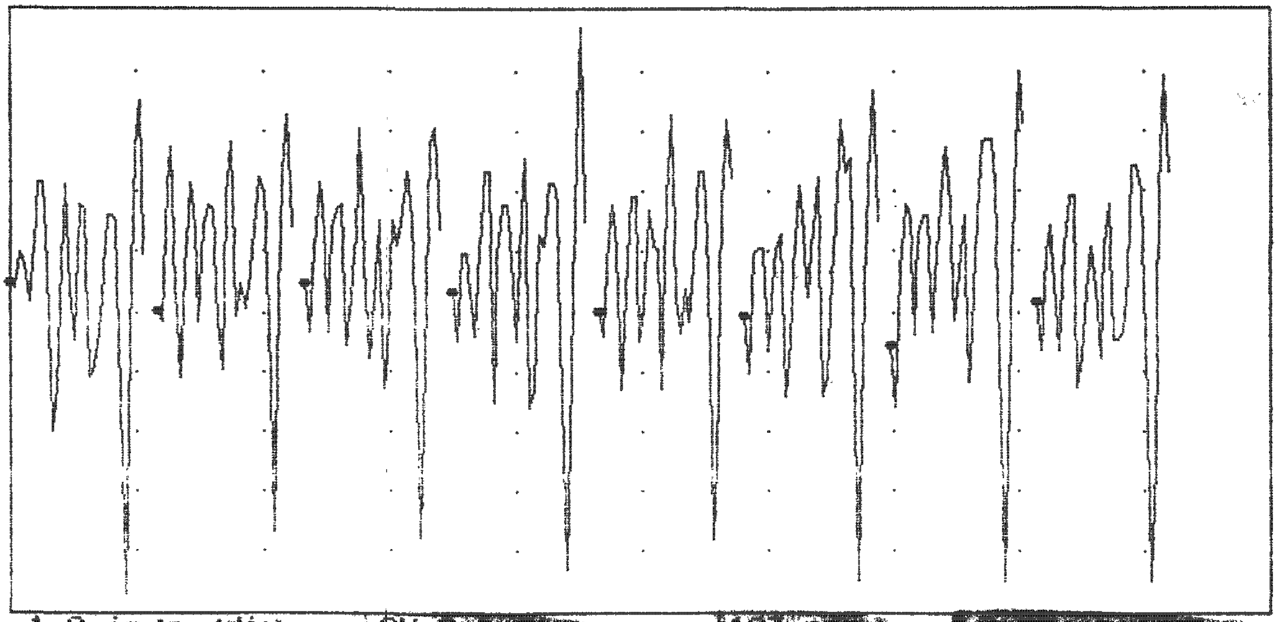
Machine: Cooling Tower Fan CH# 1 Blade # 1
16 OCT 87 08:23:29.8 UNCOMP



stet

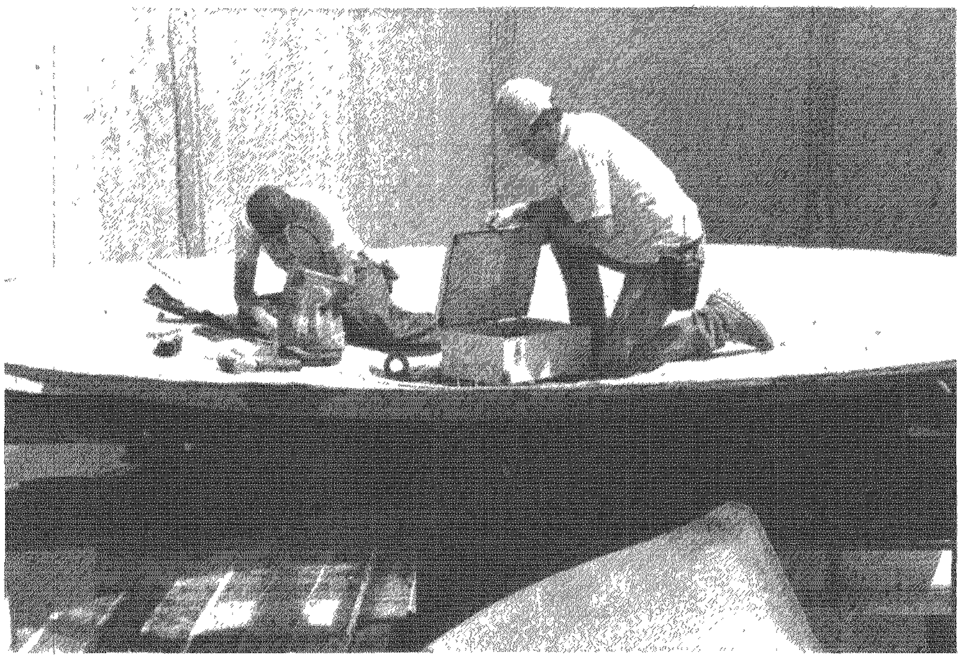
MULTIPLY * 10

Machine: Cooling Tower Fan CH# 1 Blade # 1
16 OCT 87 08:23:29.8 Steady State UNCOMP

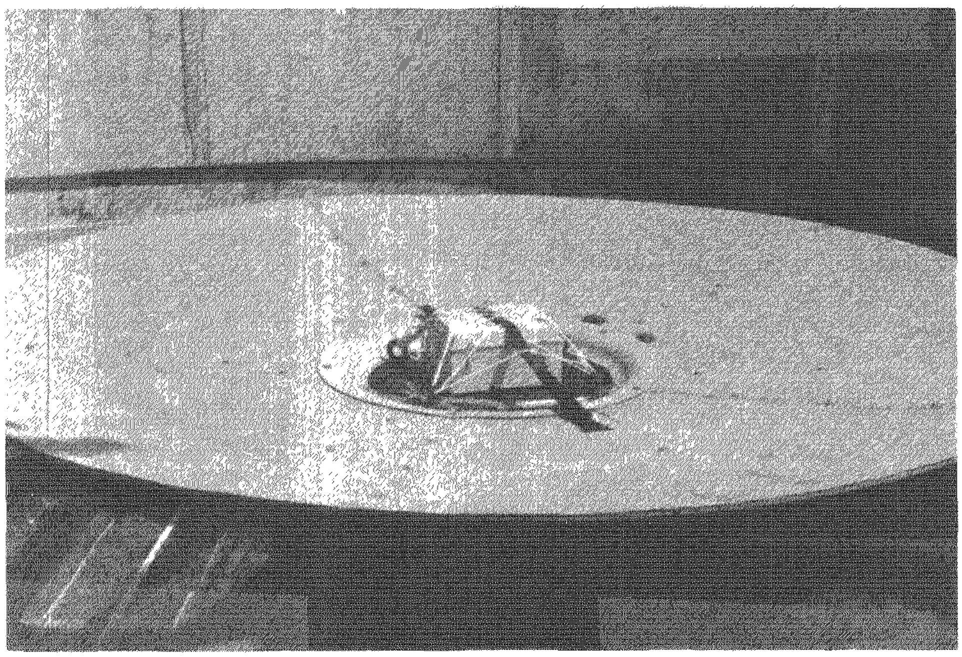


1.0 in/s /div CW [redacted] 103 [redacted] [redacted]

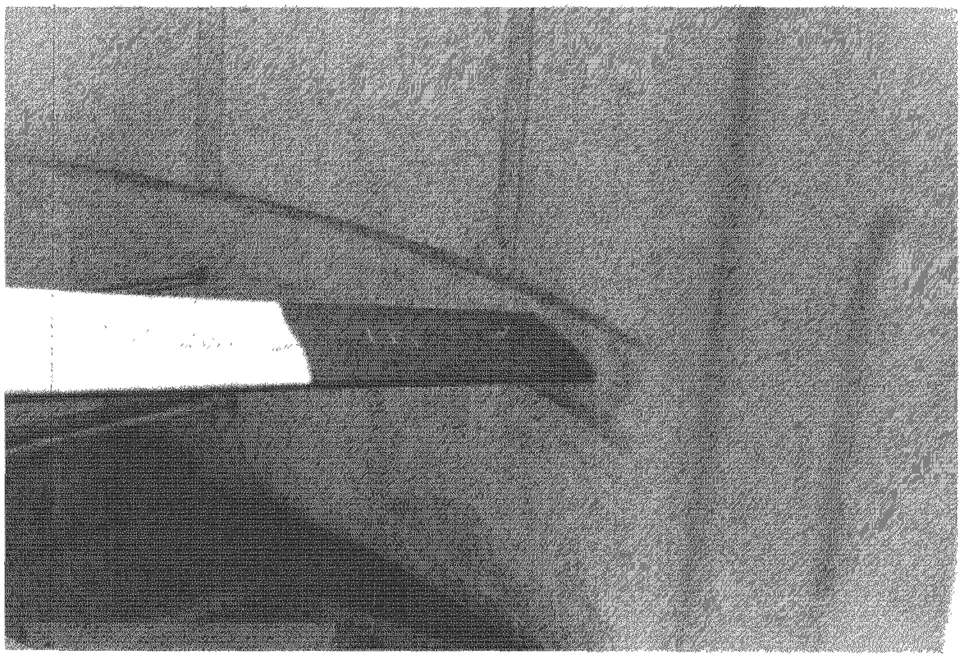
PICTURE # 4



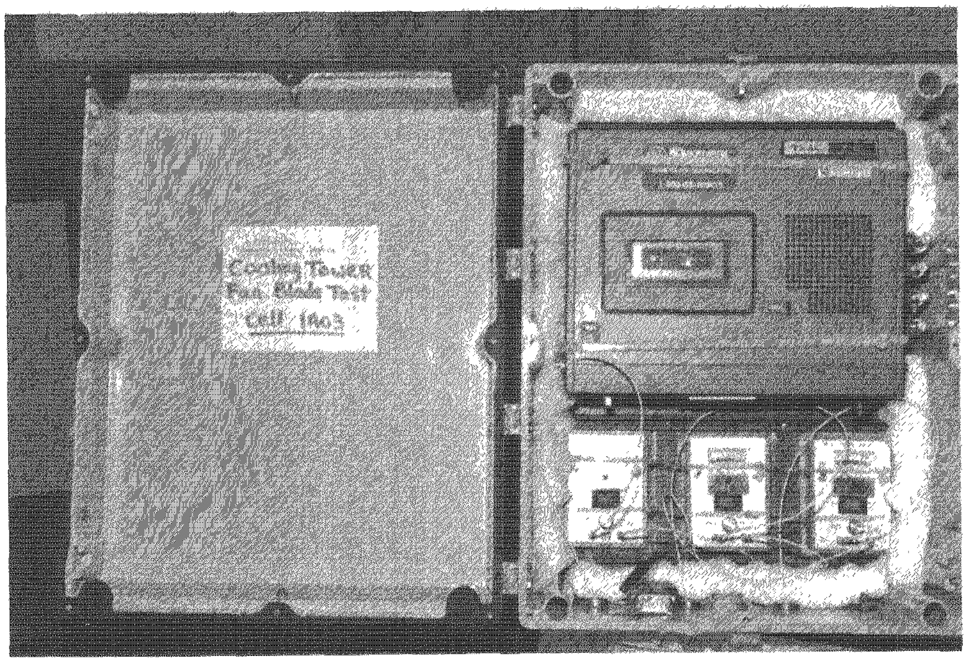
PICTURE # 5



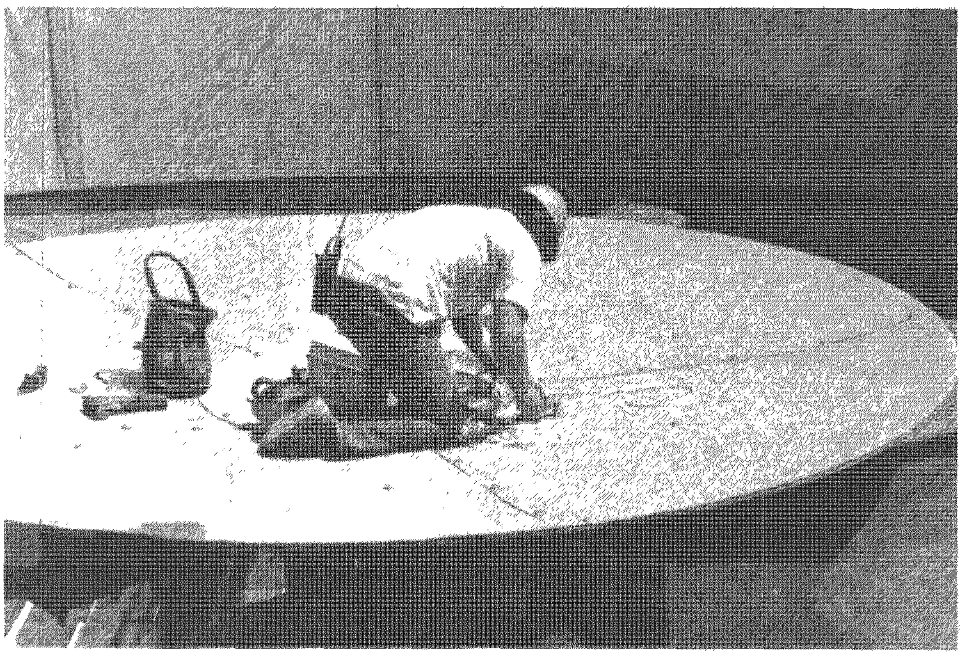
PICTURE # 1



PICTURE # 2



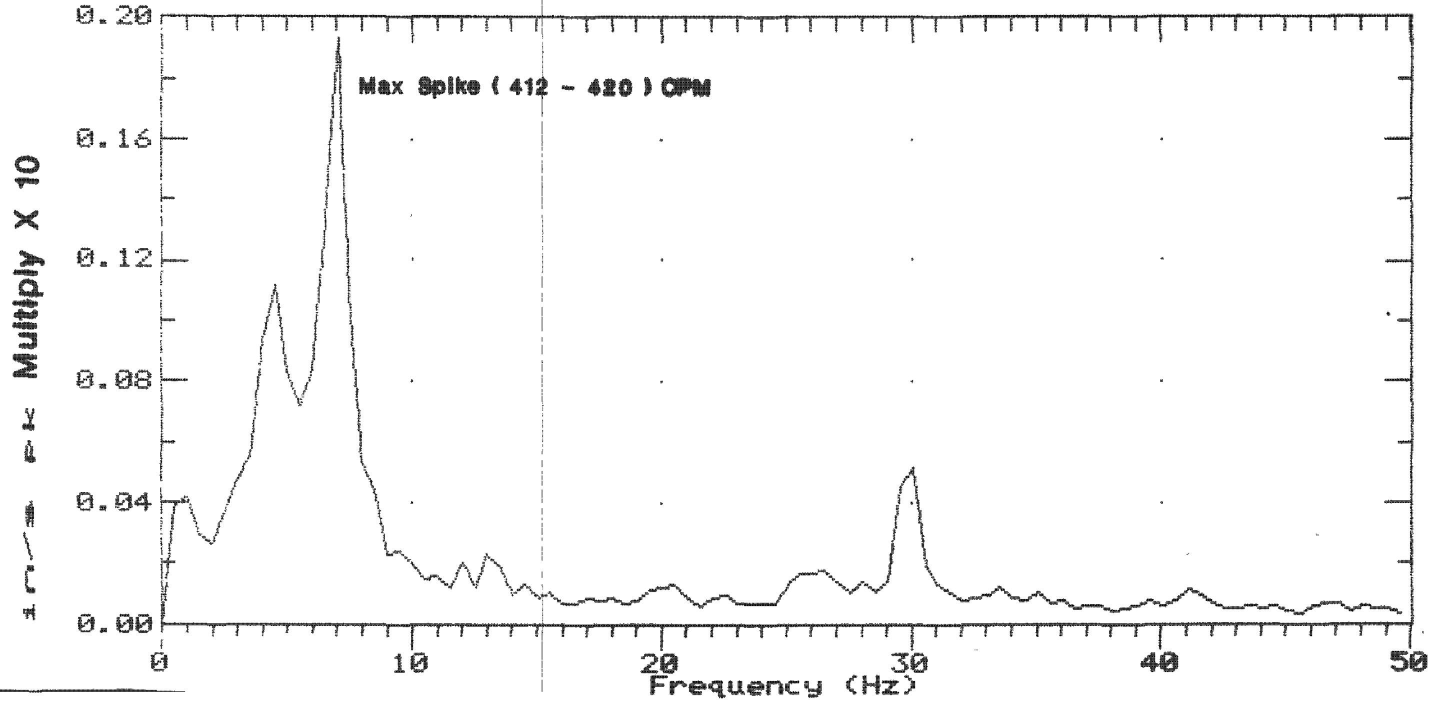
PICTURE # 3



COMPANY : IPSC ENGINEERING
PLANT : UNIT 1
JOB REFERENCE: Cooling Tower Fans
MACHINE TRAIN: 1A03 Clg Twr Fan

PLOT No.______

Machine: Cooling Tower Fan CH# 1 Blade # 1
16 OCT 87 08:14:26.9 to 16 OCT 87 08:15:46.9 UNCOMP # Ave. = 9 (spec)



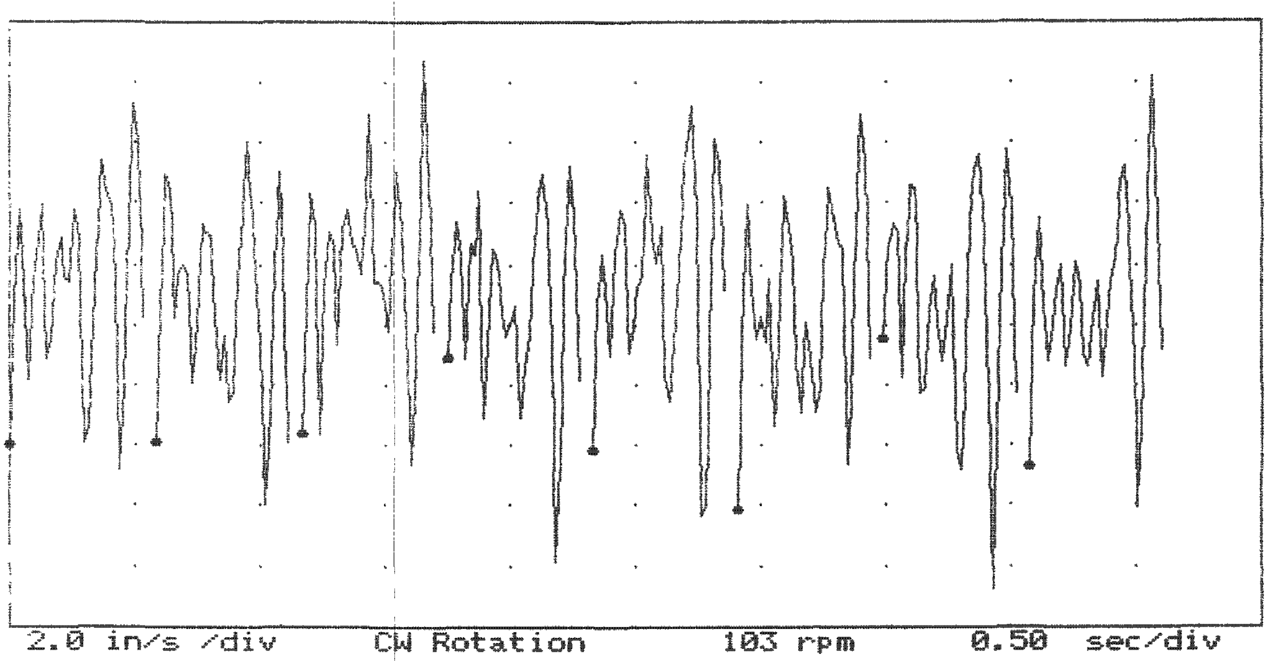
IP12_011526

ATTACHMENT # 2

COMPANY : IPSC ENGINEERING
PLANT : UNIT 1
JOB REFERENCE: Cooling Tower Fans
MACHINE TRAIN: 1A03 Clg Twr Fan

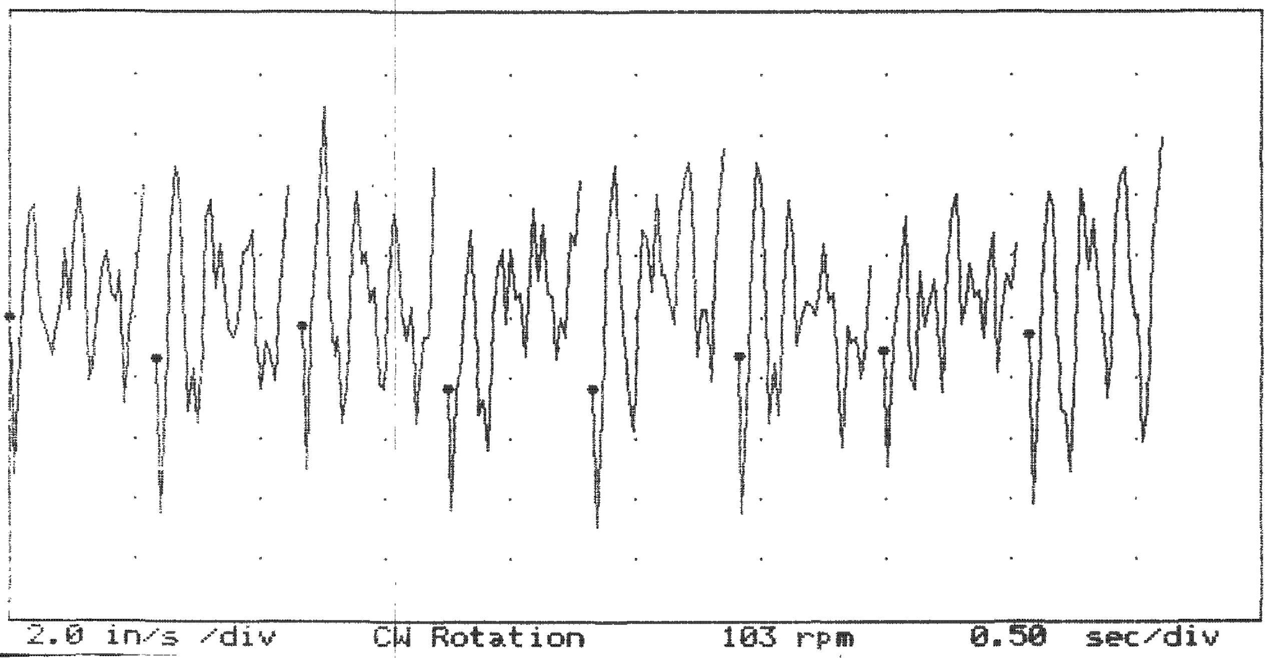
PLOT No.______

Machine: Cooling Tower Fan CH# 1 Blade # 1
16 OCT 87 08:20:49.8 Steady State UNCOMP



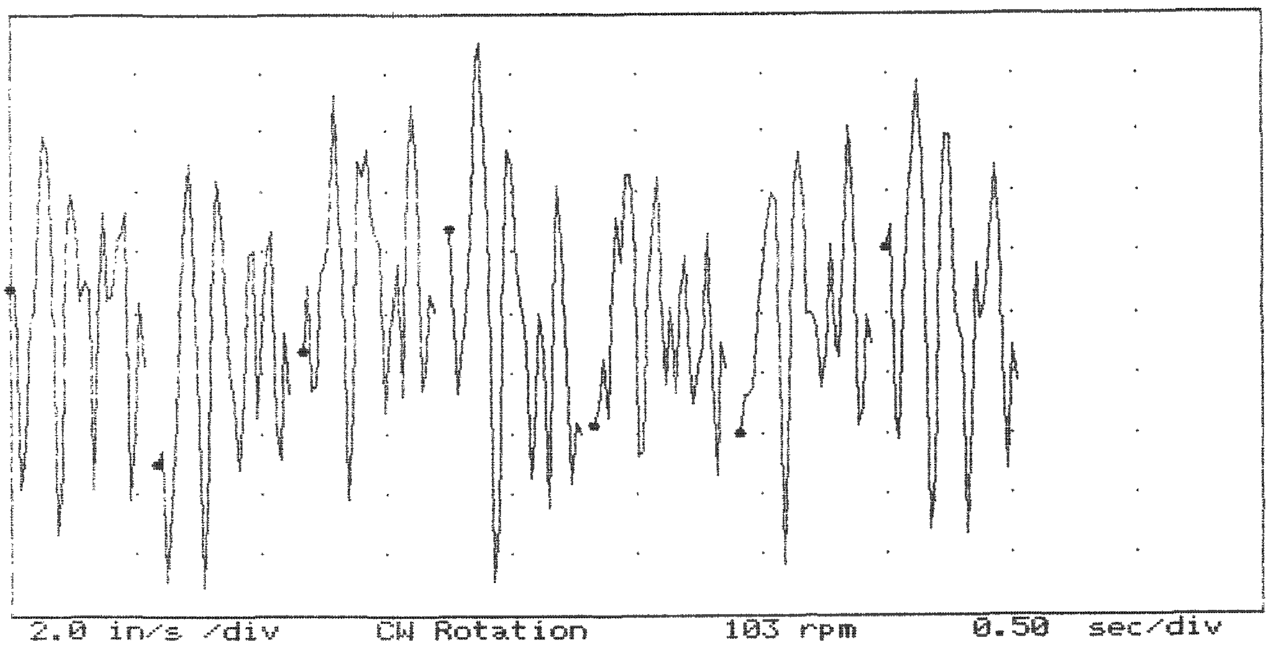
2.0 in/s /div CW Rotation 103 rpm 0.50 sec/div

Machine: Cooling Tower Fan CH# 2 Blade # 3
16 OCT 87 08:20:49.8 Steady State UNCOMP



2.0 in/s /div CW Rotation 103 rpm 0.50 sec/div

Machine: Cooling Tower Fan CH# 3 Blade # 5
16 OCT 87 08:20:49.8 Steady State UNCOMP

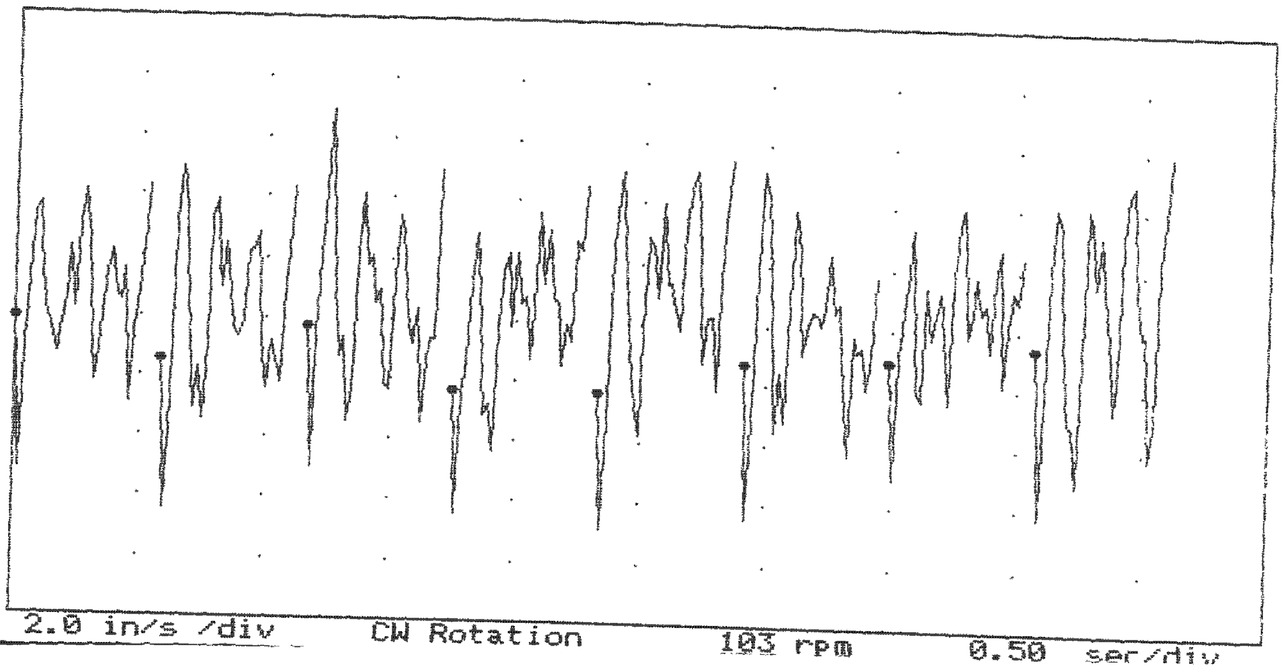


2.0 in/s /div CW Rotation 103 rpm 0.50 sec/div

COMPANY : IPSC ENGINEERING
PLANT : UNIT 1
JOB REFERENCE: Cooling Tower Fans
MACHINE TRAIN: 1A03 Clg Twr Fan

PLOT No.______

Machine: Cooling Tower Fan CH# 2 Blade # 3
16 OCT 87 08:20:49.8 Steady State UNCOMP



2.0 in/s /div CW Rotation 103 rpm 0.50 sec/div

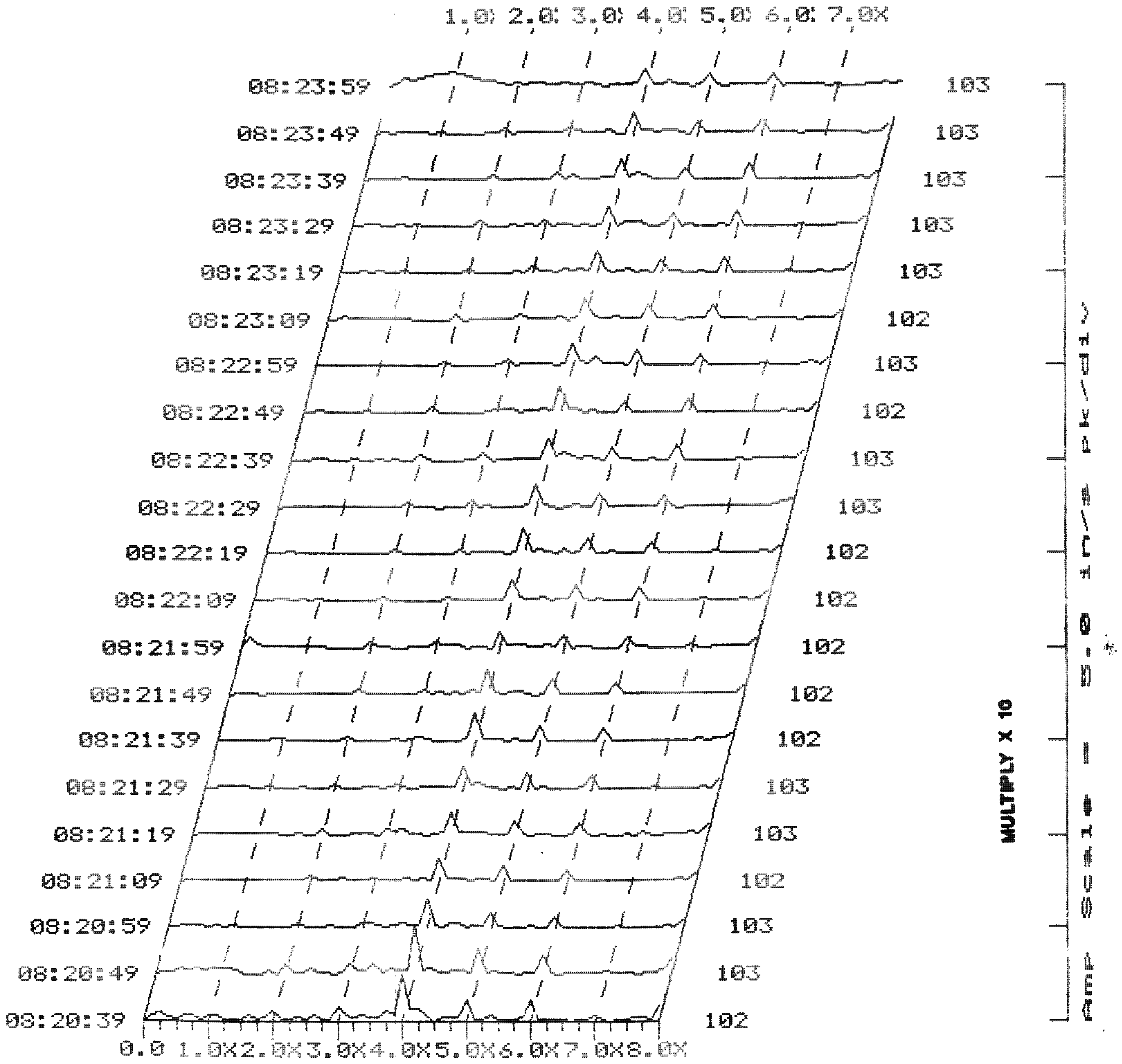
IP12_011528

ATTACHMENT # 6

COMPANY : IPSC ENGINEERING
PLANT : UNIT 1
JOB REFERENCE: Cooling Tower Fans
MACHINE TRAIN: 1A03 Clg Twr Fan

PLOT No.______

MACHINE: Cooling Tower Fan CH# 2 Blade # 3
16 OCT 87 08:20:39.8 to 16 OCT 87 08:23:59.8 Steady State UNCOMP



IP12_011529

INTERMOUNTAIN POWER SERVICE CORPORATION

January 29, 1987

File:01.03.01
14.9010
43.2600

Mr. Bruce E. Blowey
Engineer of Generation-External
LADWP
P.O. Box 111, Room 1255E
Los Angeles, CA 90051

Subject: Resonance of Cooling Tower Fan Blades may be causing fan failures.

Dear Mr. Blowey:

Attached for your information is a report covering cooling tower fan blade resonant problems, which appear to be the underlying cause for the fan failures experienced thus far at IGS.

Please contact Phil Tice at (801) 864-4414, extension 6460 for any questions or comments.

Sincerely,
Intermountain Power Service Corporation

S. Gale Chapman

S.G. Chapman
President & Chief Operations Officer

DK
WB
SGC/WB:tdt

attachment

cc: R.A. Davis
J.D. Hamblin
F. Rusk
P. Shockley

MEMORANDUM
INTERMOUNTAIN POWER SERVICE CORPORATION
JANUARY 24, 1987

TO: Dennis Killian

FROM: Wes Bloomfield

SUBJECT: Resonance excitation of cooling tower fan blades.

RE: 1B05 Fan Failure Report, 23 December, 1986.

The cooling tower fan blades are being excited at their natural resonance frequency. This excitation is probably responsible for the blade failures encountered thus far. Failures will probably continue on sub-standard blades. If failures escalate, Ceramic Cooling Tower should be requested to change the stiffness of the blades.

Each fan cell is composed of a fan with seven blades, a gear reducer mounted on a pedestal, and four concrete beams. The concrete beams stabilize the fan and gear reducer in the middle of the fan stack (Refer to fig. 1). The gear reducer input and output shaft speeds are 1785 and 103 rpm respectfully. Many vibration tests have been performed on Unit 1 & 2 fans by using the Bently Nevada (BN) absolute velocity probe mounted on the gear next to the input drive shaft (Shown on fig. 1).

On several occasions, a vibration signature of 721 cpm was measured as the most significant source of vibration (See attachment 1). A 721 cpm frequency would be considered a blade passing frequency. The location of the (BN) vibration probe, which is very directional specific, will cause an average signal, magnitude and frequency, to be read from the seven blades during each revolution of the output shaft. Therefore, seven blades at a speed of 103 rpm will produce a combined blade vibration signature of 721 cpm. It should be very clear that this signature and magnitude are representative of the blade system and not any one blade in particular. From this information it was determined that something was exciting the blades to vibrate.

In order to determine if the individual blade vibration was normal, a natural frequency or resonance test was performed on a few cooling tower fan blades in cell 2A02. Two types of transducers were used to monitor the magnitude and natural frequencies of the blades as they were excited. A low frequency absolute velocity probe was located at a position mid-span to the blade and an acceleration transducer was mounted on the tip of the blade. Both transducers were located in a vertical direction simular to the normal flow path through the cell. The velocity and acceleration measurement devices indicated a very active 1st resonance or natural frequency of 408 - 420 cpm. This test was performed several times to verify its accuracy (see attachments 2 & 3).

At first this frequency component of 408 - 420 cpm may seem trivial information, but as I will explain, it has a direct bearing on the blade failures, which IPSC has experienced thus far. The fan blades, "load up" as they perform their work. This loading up phenomenon is caused by the transportation or movement of air through the cell. This blade loaded condition is constant except when passing over one of the four concrete beam supports in the cell (Refer to fig. 1). When a blade is positioned over one of the beams, the air path is momentarily interrupted. This flow interruption actually causes a brief unloaded condition on the blade. This loss of load occurs four times on every blade for each fan revolution. Therefore, each blade is excited four times per revolution. This multiplied by a fan output shaft speed of 103 rpm will produce an excitation of 412 cpm. As can be seen, the natural designed shape of the cell will produce an excitation force, which will excite the natural frequency of the blade. This means that the blade will be excited to vibrate at its maximum amplitude. Naturally this type of excitation on the blades is not healthy. A sub-standard blade, which may have inadvertently been installed would be especially failure prone. To correct this type of problem would require changing one of the following:

1) Change the gear reducer output shaft speed. Move from 103 cpm to a faster or slower speed.

2) Change the number of concrete beam supports under the fan. Spacing must be in equal proportions.

3) Stiffen the fiberglass fan blade to shift the natural frequency to a higher value. ~~It is a common practice to laminate a leading edge on some fan blades. Hudson fan should be familiar with the process.~~ Add a foam filler in the blade which would stiffen the blade & change its natural resonance.

As you have probably already realized, the first two corrections are unrealistic at this point. The third is the only feasible alternative.

In conclusion, the blades are being excited at their maximum natural vibration mode. The number of blades on the fan has no relevance to the excitation force. The most feasible fix would be to epoxy a metallic leading edge on the blade. Hudson fan should be familiar with this process. If blade failures continue to be a problem, then Ceramic should be approached to correct this design error. Any questions or comments may be directed to Wes Bloomfield at ext. 6483.

Wes Bloomfield

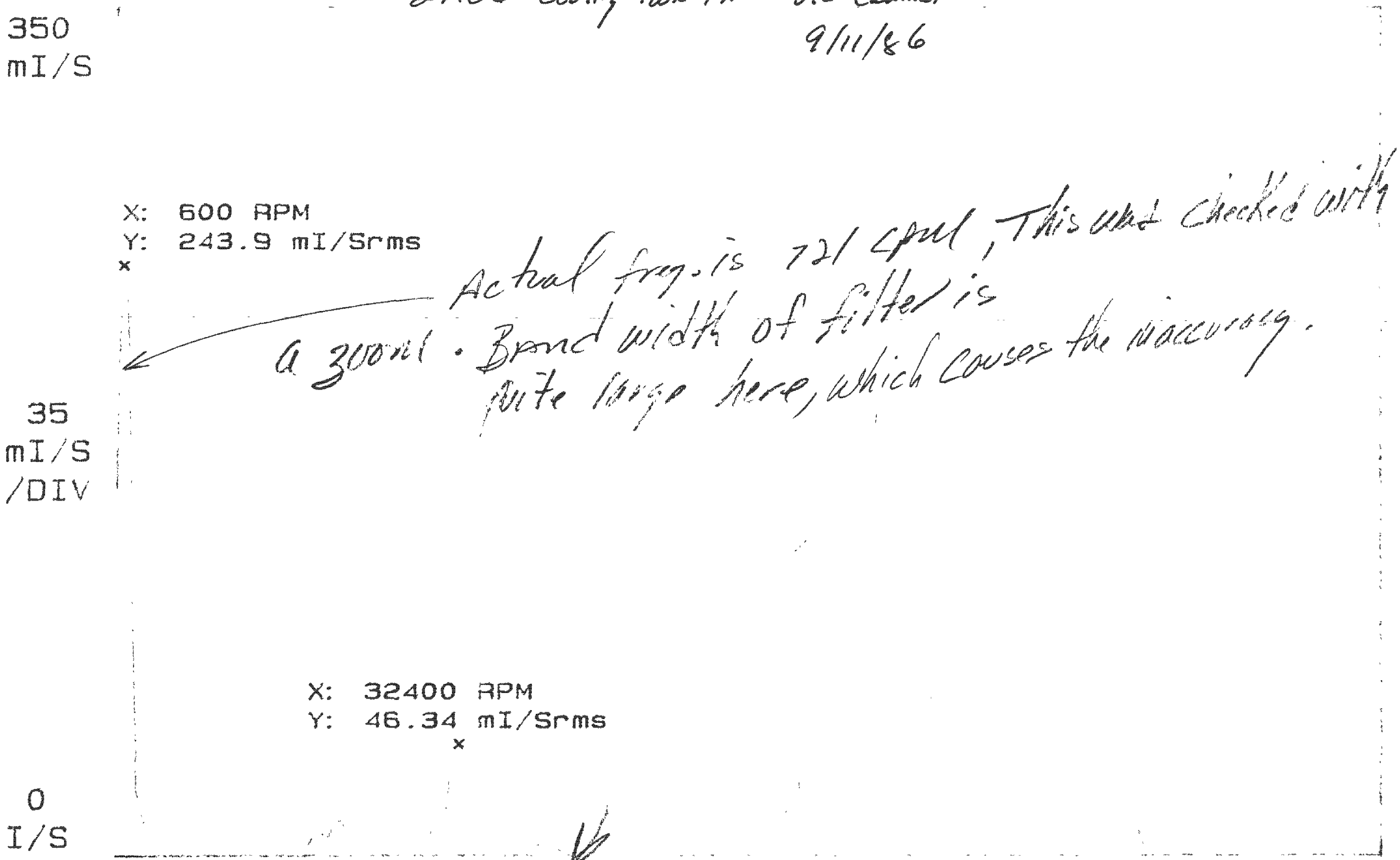
Wes Bloomfield
Reliability Engineer

cc:P.B.Tice, L.Knudsen, C.Finnegan, C.Lord, G.Rose, J.Pruett, S.Rus, D.Waters, A.Zachary

RANGE: -51 dBV STATUS: PAUSED

A: STORED 2A06 Cooling TWR FAN Vib Cabinet RMS: 10

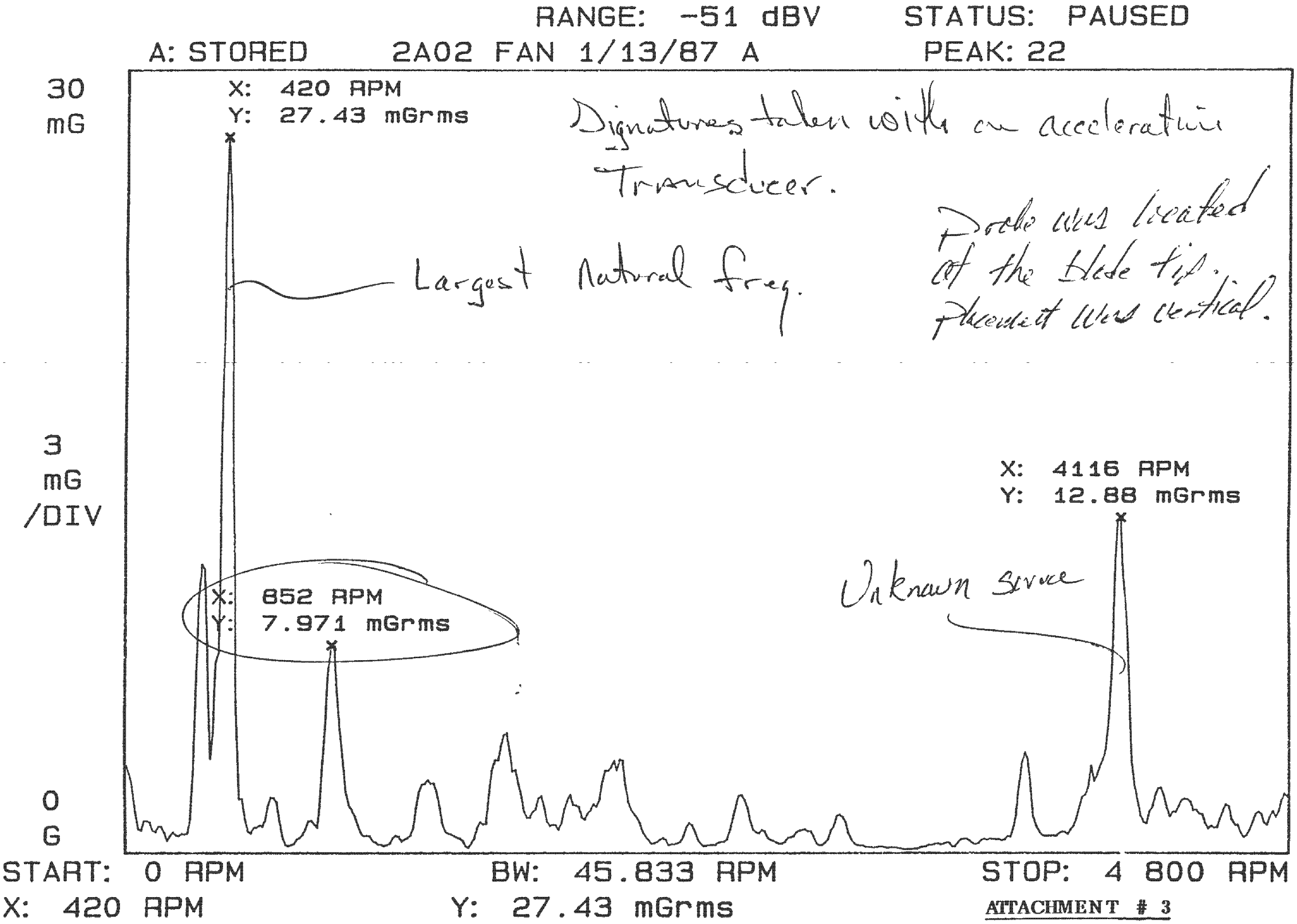
9/11/86



START: 0 RPM BW: 1 145.8 RPM STOP: 120 000 RPM

X: 600 RPM Y: 243.9 mI/Srms

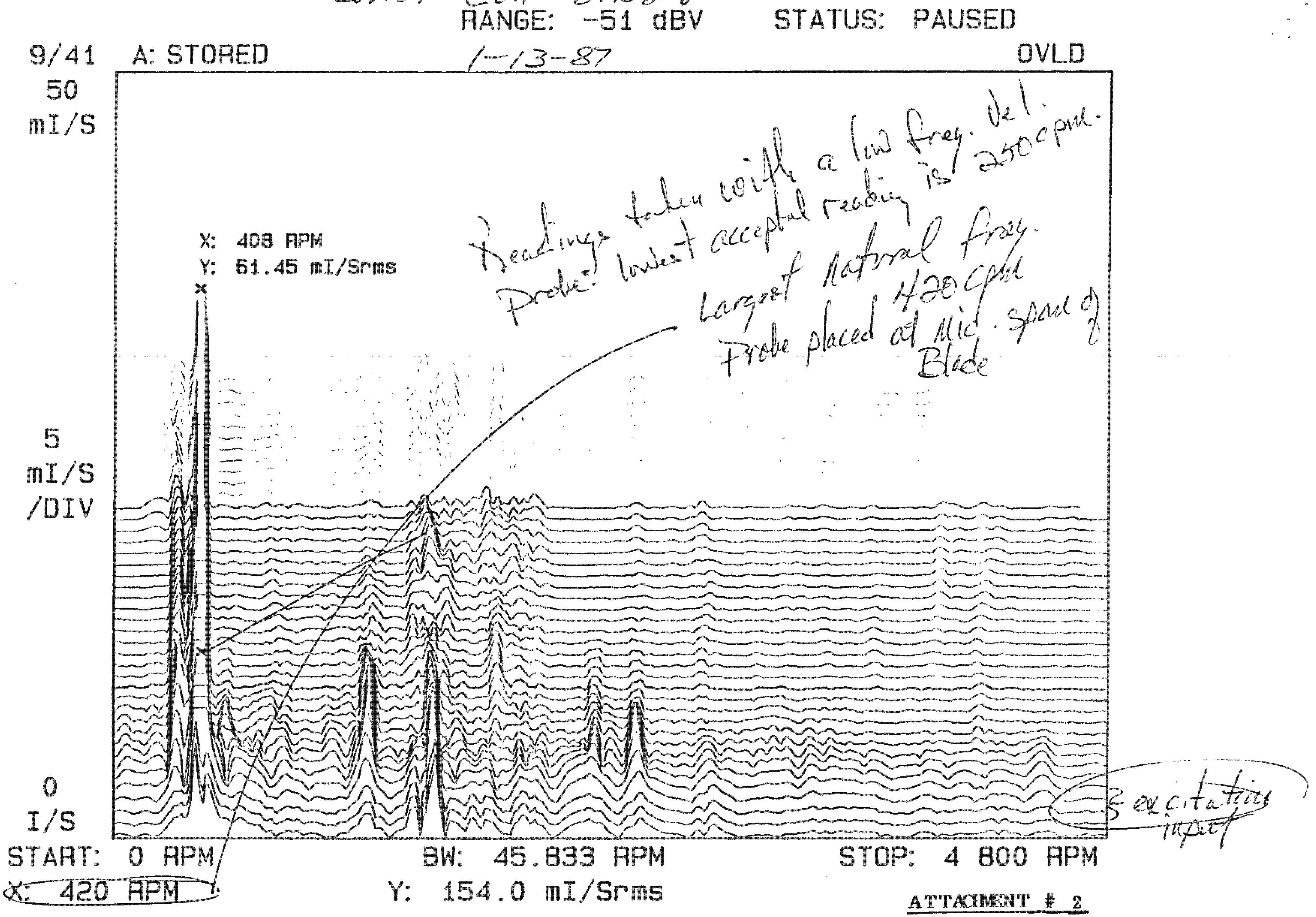
ATTACHMENT # 1



ATTACHMENT # 3

IP12_011534

Natural freq. test on Hudson 36' blades. Test was performed ~~at~~ at Cell 2A02.



ATTACHMENT # 2

Case History #3:

Fan failures traced to blade resonance excitation by the innovative use of instrumentation

Figure 1
Top view of cooling tower cell

by
Wes Bloomfield
Reliability Engineer
Intermountain Power Service Corporation
Delta, Utah

Several cooling tower fan blade failures at our plant caused us to initiate a test program to determine the vibration amplitude and various frequency components associated with our fans. As the project progressed, it became apparent that an innovative approach to instrumentation would be necessary to accurately understand and diagnose the cause of our blade failures.

The cooling tower fans at our Delta, Utah, plant have 7 blades and are 36 feet (11 metres) in diameter (Figure 1). A 200 hp (150 KW) 1785 rpm motor is located outside of each cell with a horizontal drive shaft coupled to a right angle gear box. The output shaft speed is 103 rpm.

Initial data was taken in the usual manner by attaching a Bently Nevada velocity Transducer to the gearbox (next to the input drive shaft) so that it was perpendicular to both the input and output shafts.

The vibration spectrum obtained from this transducer indicated a frequency of 721 cpm as the most significant source of vibration (Figure 2). A 721 cpm frequency would be considered a blade passing frequency because the fan rotative speed (103 rpm) multiplied by the fans seven blades produced this component. It should be noted that this frequency and magnitude are representative of the entire blades system and not any one particular blade. From this information, it was decided that something was exciting the blades.

In order to determine if the individual blade vibration was normal, a natural frequency or resonance test was performed on a few cooling tower fan blades. Two types of transducers were used to monitor the magnitudes and natural frequencies generated as the blade was stimulated. A low frequency velocity pickup was located at a position mid-span to the blade and an acceleration transducer was mounted at the blade tip. Both transducers were mounted in a vertical direction so that the sensitive axis would be in line with the flow of air through the fan.

The first set of data was gathered while the fan was in a "static" condition. Each blade was excited by pulling down on the blade until it reached approximately a 2 inch (51 mm) deflection and then the blade was released. Data from both the velocity and acceleration pickups were in agreement. This test was performed several times to insure accuracy. The Bently Nevada 108 Data Acquisition Instrument (DAI) was programmed to capture samples at 10 seconds intervals. The accelerometer signals were electronically integrated to velocity and attenuated by a factor of 10:1 within the 108 DAI. The data was then transferred to the computer by use of the Automated Diagnostics for Rotating Equipment (ADRE 3) software. The plot shown in Figure 3 was generated by averaging 9 spectrum samples within the ADRE 3 software. The averaging was used to decrease the effect of both mechanical and electronic spurious signals that may have been input to the instrumentation. As the plot indicates, the predominant frequency is in the 412-420 cpm range. This is the natural resonant frequency of the blade as mounted on the fan. At first, this frequency component may not seem significant. However, in this case, it has a direct bearing on the fan blade failures which Intermountain Power Service Corporation has experienced thus far.

In order to observe the vibratory motion of the blades with the fan running; cables were routed from the blade tip where

IP12_011536

tion of the blades with the fan running; cables were routed from the blade tip where the accelerometers were mounted to the center of the fan hub. At the fan hub, a battery operated tape recorder was attached. The tape recorder, signal conditioners, and a battery pack were installed in a NEMA 4X weatherproof housing for environmental protection. By installing the electronics at the center of the hub, the effects of centrifugal force were minimized.

A Keyphasor probe was installed to observe the gearbox output shaft. This provided a once per turn reference for the fan assembly. The ADRE 3 system was used to transfer and reduce the data from the recorded tape.

Figure 4 shows 21 samples of the running data taken at 10 second intervals. Since the Keyphasor signal was recorded simultaneously with the accelerometer signal, this plot was labeled in terms of multiples of running speed, commonly referred to as "X" (e.g. 1X, 2X, etc.). The data taken at 08:23:29 was selected as being typical of the average of the 21 spectrums (Figure 5). This plot indicates a high 4X component which was found to be related to the fan cell design.

The design of the fan cell is such that in the normal operating mode, the fan blades "load up" as they perform work. This loading up is caused by the movement of air through the cell. The loaded blade condition is constant except when the blade passes over one of the four concrete support beams in the cell. When a blade is positioned over one of the beams, the air path is momentarily interrupted. This flow interruption actually causes a brief unloaded condition on the blade. This loss of load occurs four times per fan revolution on each blade. Therefore, each blade was excited at 4 times the fan speed of 103 rpm, which produced the 4X frequency of 412 cpm.

As can be seen, the fan cell design provided an excitation force which excited the natural frequency of the blades. This meant that the excited blade vibrated at a high amplitude. Naturally, this type of blade excitation was not healthy. A sub-standard blade, which could have been installed inadvertently, would be especially failure prone. The obvious solution was to change the resonant frequency of the blades. This could be accomplished by filling each blade with foam, or by constructing the blades with a stiffer, fiberglass configuration. Either ►

Figure 2
Original spectrum of a 7 bladed fan showing a high blade passing frequency

Figure 3
Spectrum of an individual blade natural resonance frequency

Figure 4
Waterfall diagram showing spectrum samples taken at 10 second intervals

Figure 5
The high 4x frequency component was found to be related to the fan cell design.

Figure 6
Timebase signal of the blade mounted accelerometer. Note: arrows indicate when blade is over the thicker support beam.

Figure 7
Spectrum with the new 10 blade design.

of these solutions would increase the natural frequency. Since each solution would have required extensive rework by the manufacturer, further investigation into the problem was conducted.

It was discovered by observing the integrated accelerometer time base signals and frequency spectrums that a reverse (against rotation) "wobble" of the fan blade hub was occurring. As can be seen in Figure 1, a blade that was positioned over the beam and in an unloaded condition would be followed .02 seconds later by the next blade to become unloaded, approximately 103 degrees against rotation, from the previously unloaded blade. This phenomenon made the hub wobble *against* the rotation of the fan.

It was also noted that each fourth cycle of vibration on the time base signal was of greater amplitude (Figure 6). This was found due to the concrete support directly under the drive shaft, which is twice as thick as the other three supporting beams. This information, along with velocity profiles across the egress of the fan, indicated a severe lack of performance. As the fan operated, it lifted the air but, at the same time it pushed air ahead of the blade. This volume of air caused the blade to lift even higher than when passing over the other three beams. This phenomenon was a great surprise to the designer!

In an attempt to solve the problem on site, hub rings have been drilled so that 10 blades could be installed. The effect of the 10 blade configuration is such that there are always two opposite blades unloaded at the same time. The relative vibration amplitudes have decreased with the 10 blade configuration as can be seen in Figure 7.

We are currently encouraging the tower manufacturer, due to performance compliance, and vibration maintenance considerations, to upgrade to the 10 blade design for all of our fans. In the meantime, the manufacturer of the blades is continuing to test fiberglass layup configurations to change the blade resonance frequency.

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